Low Emissions Combustion Engines for Motor Vehicles

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During the past 10-15 years, very significant advances in controlling exhaust emissions from automobile power plants have been made. Initially, emissions reductions were achieved through careful readjustment and control of engine operating conditions (1). More recently, highly effective exhaust treatment devices requiring a minimum of basic modification to the already highly developed internal combustion engine have been demonstrated. These are based on thermal and/or catalytic oxidation of hydrocarbons (HC) and carbon monoxide (CO) in the engine exhaust system (2,3,4). Nitrogen oxide (NO_x) emissions have been reduced to some extent through a combination of retarded ignition timing and exhaust gas recirculation (EGR), both factors serving to diminish severity of the combustion process temperature-time history without substantially altering design of the basic engine (5).

Basic combustion process modification as an alternative means for emissions control has received less attention than the foregoing techniques, though it has been demonstrated that certain modified combustion systems can in principle yield significant pollutant reductions without need for exhaust treatment devices external to the engine. Additionally, it has been demonstrated that when compared with conventional engines controlled to low emissions levels, modified combustion processes can offer improved fuel economy.

Nearly all such modifications involve engine designs permitting combustion of fuel-air mixtures lean beyond normal ignition limits. As will be shown, decreased mean combustion temperatures associated with extremely lean combustion tend to limit the rate of nitric oxide (NO) formation and, hence, the emission of NO_x. At the same time, the relatively high oxygen content of lean mixture combustion products tends to promote complete oxidation of unburned HC and CO provided that combustion gas temperatures are sufficiently high during late portions of the engine cycle.

The purpose of this paper is to present an overall review of the underlying concepts and current status of unconventional engines employing modified combustion as a means for emissions control. Detailed findings related to specific power plants or to specific applications will be treated by the papers which follow.

Throughout the paper exhaust emissions will be compared with emissions standards legislated for the years 1975 and 1976. As a result of Environmental Protection Agency (EPA) actions suspending the 1975 HC and CO standards and the 1976 NO_x standard, several sets of values exist. These are listed in Table I and in the text will be referenced either as statutory (original standards as set by the Clean Air Act Amendment of 1970) or as interim standards as set by the EPA.

Theoretical Basis for
Combustion Modification

Figure 1 has been derived from experimental measurements (6) of the rate of NO formation in combustion processes under conditions typical of engine operation. This figure demonstrates two major points related to control of NO_x emissions: First, the slow rate of NO formation relative to the rates of major combustion reactions responsible for heat release and, second, the strong influence of fuel-air equivalence ratio on the rate of NO formation.

Experimental combustion studies (7) employing "well-stirred reactors" have shown that hydrocarbon-air combustion rates can be correlated by an expression of the form

$$\frac{N}{V\,p^{1.8}} = 48\,\frac{\text{Gram-Moles/Liter-Second}}{\text{Atm}^{1.8}}$$

where:

N = moles reactants consumed per second
V = combustion volume
p = total pressure

For conditions typical of engine operation, this expression yields a time of approximately 0.1 ms for completion of major heat release reactions following ignition of a localized parcel of fuel-air mixture within the combustion chamber. Comparison with Figure 1 shows that the time required for formation of significant amounts of NO in combustion gases is at least a factor of 10 greater. Thus, in principle, energy conversion can be effected in times much shorter than required for NO formation. In the conventional spark ignition engine, the relatively lengthy flame travel process permits combustion products to remain at high temperatures sufficiently long that considerable NO formation occurs.

Figure 2, which consolidates the data of Figure 1, indicates that maximum rates of NO formation are observed at fuel-air equivalence ratios around 0.9 (fuel lean). For richer mixtures, the concentrations of atomic and diatomic oxygen, which participate as reactants in the formation of NO in combustion gases, decrease. On the other hand, for mixtures leaner than approximately 0.9 equivalence ratio, decreasing combustion temperatures result in lower NO formation rates.

Figure 2 serves as a basis for combustion process modification. Operation with extremely rich fuel-air mixtures (Point A of Figure 2), of course, results in low NO_x emissions since the maximum chemical equilibrium NO level is greatly reduced under such conditions. However, the resultant penalties in terms of impaired fuel economy and excessive HC and CO emissions are well known. An alternative is operation with extremely lean mixtures (Point B), lean beyond normal ignition limits. Combustion under such conditions can lead to low NO_x emissions while at the same time providing an excess of oxygen for complete combustion of CO and HC.

Operation of internal combustion engines with extremely lean overall fuel-air ratios has been achieved in several ways, employing a number of differing combustion chamber configurations. One approach involves ignition of a very small and localized quantity of fuel-rich and ignitable mixture (Point A of Figure 2), which in turn serves to inflame a much larger quantity of surrounding fuel-air mixture too lean for ignition under normal circumstances. The bulk or average fuel-air ratio for the process corresponds to Point B of Figure 2; and, as a consequence, reduced exhaust emissions should result.

A second approach involves timed staging of the combustion process. An initial rich mixture stage in which major combustion reactions are carried out is followed by extremely rapid mixing of rich mixture combustion products with dilution air. The transition from initial Point A to final Point B in Figure 2 is, in principle, sufficiently rapid that little opportunity for NO formation exists. Implicit here is utilization of the concept that the heat release reactions involved in the transition from Point A to Point B can be carried out so rapidly that time is not available for formation of significant amounts of NO.

Reciprocating spark ignition engines designed to exploit the foregoing ideas are usually called stratified charge engines, a term generally applied to a large number of designs encompassing a wide spectrum of basic combustion processes.

Open-Chamber Stratified Charge Engines

Stratified charge engines can be conveniently divided into two types: open-chamber and dual-chamber engines. The open-chamber stratified charge engine has a long history of research interest. Those engines reaching the most advanced stages of development are probably the Ford-programmed combustion process (PROCO) (8) and Texaco's controlled combustion process (TCCS) (9). Both engines employ a combination of inlet air swirl and direct timed combustion chamber fuel injection to achieve a local fuel-rich ignitable mixture near the point of ignition. The overall mixture ratio under most operating conditions is fuel lean.

The Texaco TCCS engine is illustrated schematically in Figure 3. During the engine inlet stroke, an unthrottled supply of air enters the cylinder through an inlet port oriented to promote a specified level of air swirl within the cylinder and combustion chamber. As the subsequent compression stroke nears completion, fuel is injected into and mixes with an element of swirling air charge. This initial fuel-air mixture is spark ignited, and a flame zone is established downstream from the nozzle. As injection continues, fuel-air mixture is continuously swept into the flame zone. The total quantity of fuel consumed per cycle and, hence, engine power output, are controlled by varying the duration of fuel injection. Under nearly all engine operating conditions, the total quantity of fuel injected is on the lean side of stoichiometric. The TCCS system has been under development by Texaco since the 1940's. To date, this work has involved application of the process to a wide variety of engine configurations.

Like the TCCS engine, the PROCO system (Figure 4) employs timed combustion chamber fuel injection. However, in contrast to the TCCS system, the PROCO system is based on formation of a premixed fuel-air mixture prior to ignition. Fuel injection and inlet air swirl are coordinated to provide a small portion of rich mixture near the point of ignition surrounded by a large region of increasingly fuel-lean mixture. Flame propagation proceeds outward from the point of ignition through the leaner portions of the combustion chamber.

Both the TCCS and PROCO engines are inherently low emitters of CO, primarily a result of lean mixture combustion. Unburned HC and NO_x emissions have been found to be lower than those typical of uncontrolled conventional engines, but it appears that additional control measures are required to meet statutory 1976 Federal emissions standards.

The U.S. Army Tank Automotive Command has sponsored development of low emissions TCCS and PROCO power plants for light-duty Military vehicles. These power plants have been based on conversion of the 4-cylinder, 70-hp L-141 Jeep engine. The vehicles in which these engines were placed were equipped with oxidizing catalysts for added control of HC and CO emissions, and EGR was used as an additional measure for control of NO_x.

Results of emissions tests on Military Jeep vehicles equipped with TCCS and PROCO engines are listed in Table II (10). At low mileage these vehicles met the statutory 1976 emissions standards. Deterioration problems related to HC emission would be expected to be similar to those of conventional engines equipped with oxidizing catalysts. This is evidenced by the increase in HC emissions with mileage shown by Table II. NO_x and CO emissions appear to have remained below 1976 levels with mileage accumulation.

Table III presents emissions data at low mileage for several passenger car vehicles equipped with PROCO engine conversions (10). These installations included noble metal catalysts and EGR for added control of HC and NO_x emissions, respectively. All vehicles met the statutory 1976 standards at low mileage. Fuel consumption data, as shown in Table III, appear favorable when contrasted with the fuel economy for current production vehicles of similar weight.

Fuel requirements for the TCCS and PROCO engines differ substantially. The TCCS concept was initially developed for multifuel capability; as a consequence, this engine does not have a significant octane requirement and is flexible with regard to fuel requirements. In the PROCO engine combustion chamber, an end gas region does exist prior to completion of combustion; and, as a consequence, this engine has a finite octane requirement.

Prechamber Stratified Charge Engines

A number of designs achieve charge stratification through division of the combustion region into two adjacent chambers. The emissions reduction potential for two types of dual-chamber engines has been demonstrated. First, in a design traditionally called the "prechamber engine," a small auxiliary or ignition chamber equipped with a spark plug communicates with the much larger main combustion chamber located in the space above the piston (Figure 5). The prechamber typically contains 5-15% of the total combustion volume. In operation of this type of engine, the prechamber is supplied with a small quantity of fuel-rich ignitable fuel-air mixture while a very lean and normally unignitable mixture is supplied to the main chamber above the piston. Expansion of high temperature flame products from the prechamber leads to ignition and burning of the lean main chamber fuel-air charge.

The prechamber stratified charge engine has existed in various forms for many years. Early work by Ricardo (11) indicated that the engine could perform very efficiently within a limited range of carefully controlled operating conditions. Both fuel-injected and carbureted prechamber engines have been built. A fuel-injected design initially conceived by Brodersen (12) was the subject of extensive study at the University of Rochester for nearly a decade (13,14). Unfortunately, the University of Rochester work was undertaken prior to widespread recognition of the automobile emissions problem; and, as a consequence, emissions characteristics of the Brodersen engine were not determined. Another prechamber engine receiving attention in the early 1960's is that conceived by R. M. Heintz (15). The objectives of this design were reduced HC emissions, increased fuel economy, and more flexible fuel requirements.

Initial experiments with a prechamber engine design called "the torch ignition engine" were reported in the U.S.S.R. by Nilov (16) and later by Kerimov and Mekhtier (17). This designation refers to the torchlike jet of hot combustion gases issuing from the precombustion chamber upon ignition. In the Russian designs, the orifice between prechamber and main chamber is sized to produce a high velocity jet of combustion gases. In a recent publication (18), Varshaoski et al. have presented emissions data obtained with a torch engine system. These data show significant pollutant reductions relative to conventional engines; however, their interpretation in terms of requirements based on the Federal emissions test procedure is not clear.

A carbureted three-valve prechamber engine, the Honda Compound Vortex-Controlled Combustion (CVCC) system, has received considerable recent publicity as a potential low emissions power plant (19). This system is illustrated schematically in Figure 6. Honda's current design employs a conventional engine block and piston assembly. Only the cylinder head and fuel inlet system differ from current automotive practice. Each cylinder is equipped with a small precombustion chamber communicating by means of an orifice with the main combustion chamber situated above the piston. A small inlet valve is located in each prechamber. Larger inlet and exhaust valves typical of conventional automotive practice are located in the main combustion chamber. Proper proportioning of fuel-air mixture between prechamber and main chamber is achieved by a combination of throttle control and appropriate inlet valve timing. Inlet ports and valves are oriented to provide specific levels of air swirl and turbulence in the combustion chamber. In this way, a relatively slow and uniform burning process giving rise to elevated combustion temperatures late in the expansion stroke and during the exhaust process is achieved. High temperatures in this part of the engine cycle are necessary to promote complete oxidation of HC and CO. It should be noted that these elevated temperatures are necessarily obtained at the expense of a fuel economy penalty.

Results of emissions tests with the Honda engine have been very promising. The emissions levels shown in Table IV for a number of lightweight Honda Civic vehicles are typical and demonstrate that the Honda engine can meet statutory 1975-1976 HC and CO standards and can approach the statutory 1976 NO_x standard (10). Of particular importance, durability of this system appears excellent as evidenced by the high mileage emissions levels reported in Table IV. The noted deterioration of emissions after 30,000-50,000 miles of engine operation was slight and apparently insignificant.

Recently, the EPA has tested a larger vehicle converted to the Honda system (20). This vehicle, a 1973 Chevrolet Impala with a 350-CID V-8 engine, was equipped with cylinder heads and induction system of Honda manufacture. Test results are presented in Table V for low vehicle mileage. The vehicle met the present 1976 interim Federal emissions standards though NO_x levels were substantially higher than for the much lighter weight Honda Civic vehicles.

Fuel economy data indicate that efficiency of the Honda engine, when operated at low emissions levels, is somewhat poorer than that typical of well-designed conventional engines operated without emissions controls. However, EPA data for the Chevrolet Impala conversion show that efficiency of the CVCC engine meeting 1975-1976 interim standards was comparable to or slightly better than that of 1973 production engines of similar size operating in vehicles of comparable weight. It has been stated by automobile manufacturers that use of exhaust oxidation catalysts beginning in 1975 will result in improved fuel economy relative to 1973 production vehicles. In this event fuel economy of catalyst-equipped conventional engines should be at least as good as that of the CVCC system.

The apparent effect of vehicle size (more precisely the ratio of vehicle weight to engine cubic inch displacement) on NO_x emissions from the Honda engine conversions demonstrates the generally expected response of NO_x emissions to increased specific power demand from this type of engine. For a given engine cubic inch displacement, maximum power output can be achieved only by enriching the overall fuel-air mixture ratio to nearly stoichiometric proportions and at the same time advancing ignition timing to the MBT point. Both factors give rise to increased NO_x emissions. This behavior is evidenced by Table VI, which presents steady state emissions data for the Honda conversion of the Chevrolet Impala (20). At light loads, NO_x emissions are below or roughly comparable to emissions from a conventionally powered 1973 Impala. This stock vehicle employs EGR to meet the 1973 NO_x standard. It is noted in Table VI that for the heaviest load condition reported, the 60-mph cruise, NO_x emissions from the Honda conversion approached twice the level of emissions from the stock vehicles. This points to the fact that in sizing engines for a specific vehicle application, the decreased air utilization (and hence specific power output) of the prechamber engine when operated under low emissions conditions must be taken into consideration.

Divided-Chamber Staged Combustion Engine

Dual-chamber engines of another type, often called "divided-chamber" or "large-volume prechamber" engines, employ a two-stage combustion process. Here initial rich mixture combustion and heat release (first stage of combustion) are followed by rapid dilution of combustion products with relatively low temperature air (second stage of combustion). In terms of the concepts previously developed, this process is initiated in the vicinity of Point A of Figure 2. Subsequent mixing of combustion products with air is represented by a transition from Point A to Point B. The object of this engine design is to effect the transition from Point A to Point B with sufficient speed that time is not available for formation of significant quantities of NO. During the second low temperature stage of combustion (Point B), oxidation of HC and CO goes to completion.

An experimental divided-chamber engine design that has been built and tested is represented schematically in Figure 7 (21,22). A dividing orifice (3) separates the primary combustion chamber (1) from the secondary combustion chamber (2), which includes the cylinder volume above the piston top. A fuel injector (4) supplies fuel to the primary chamber only. Injection timing is arranged such that fuel continuously mixes with air entering the primary chamber during the compression stroke. At the end of compression, as the piston nears its top center position, the primary chamber contains an ignitable fuel-air mixture while the secondary chamber adjacent to the piston top contains only air. Following ignition of the primary chamber mixture by a spark plug (6) located near the dividing orifice, high temperature rich mixture combustion products expand rapidly into and mix with the relatively cool air contained in the secondary chamber. The resulting dilution of combustion products with attendant temperature reduction rapidly suppresses formation of NO. At the same time, the presence of excess air in the secondary chamber tends to promote complete oxidation of HC and CO.

Results of limited research conducted both by university and industrial laboratories indicate that NO_x reductions of as much as 80-95% relative to conventional engines are possible with the divided-chamber staged combustion process. Typical experimentally determined NO_x emissions levels are presented in Figure 8 (23). Here NO_x emissions for two different divided-chamber configurations are compared with typical emissions levels for conventional uncontrolled automobile engines. The volume ratio, β, appearing as a parameter in Figure 8, represents the fraction of total combustion volume contained in the primary chamber. For β values approaching 0.5 or lower, NO_x emissions reach extremely low levels. However, maximum power output capability for a given engine size decreases with decreasing β values. Optimum primary chamber volume must ultimately represent a compromise between low emissions levels and desired maximum power output.

HC and particularly CO emissions from the divided-chamber engine are substantially lower than conventional engine levels. However, further detailed work with combustion chamber geometries and fuel injection systems will be necessary to fully evaluate the potential for reduction of these emissions. Table VII presents results of tests cited by the National Academy of Sciences (10).

Emissions from the divided-chamber engine are compared with those from a laboratory PROCO stratified charge engine, the comparison being made at equal levels of NO_x emissions. NO_x emissions were controlled to specific levels by addition of EGR to the PROCO engine and by adjustment of operating parameters for the divided-chamber engine. These data indicate that the divided-chamber engine is capable of achieving very low NO_x emissions with relatively low HC and CO emissions.

As shown by Table VII, fuel economy of the divided-chamber staged combustion engine is comparable to that of conventional piston engines without emissions controls. When compared with conventional piston engines controlled to equivalent low NO_x emissions levels, the divided-chamber engine is superior in terms of fuel economy.

<u>The Diesel Engine</u>

The diesel engine can be viewed as a highly developed form of stratified charge engine. Combustion is initiated by compression ignition of a small quantity of fuel-air mixture formed just after the beginning of fuel injection. Subsequently, injected fuel is burned in a heterogeneous diffusion flame. Overall fuel-air ratios in diesel engine operation are usually extremely fuel lean. However, major combustion reactions occur locally in combustion chamber regions containing fuel-air mixtures in the vicinity of stoichiometric proportions.

The conventional diesel engine is characterized by low levels of CO and light HC emissions, a result of lean mixture operation. On a unit power output basis, NO_x emissions from diesel engines are typically lower than those of uncontrolled gasoline engines, a combined result of diffusion combustion and, in an approximate sense, low mean combustion temperatures. Work devoted to mathematical simulation of diesel combustion has shown that NO formation occurs primarily in combustion products formed early in the combustion process, with the later portions of diffusion-controlled combustion contributing substantially less (24).

Table VIII presents emissions levels for three diesel-powered passenger cars as reported by the EPA (25). These vehicles, a Mercedes 220D, Opel Rekord 2100D, and Peugeot 504D, were powered by 4-cylinder engines ranging in size from 126-134 CID with power ratings ranging from 65-68 bhp. Two of the diesel-powered vehicles were capable of meeting the 1975 statutory emissions standards. NO_x emissions were in excess of the original Federal 1976 standards but were well within present interim standards.

The preceding data do not include information on particulate and odorant emissions, both of which could be important problems with widespread diesel engine use in automobiles. Complete assessment of the environmental potential for the diesel engine would have to include consideration of these factors as well as emission of polynuclear aromatic hydrocarbons. All are the subject of ongoing research.

Fuel economy data referred to both 1972 and 1975 Federal test procedures are presented in Table VIII. As expected, diesel engine fuel economies are excellent when compared with gasoline engine values. However, a more accurate appraisal would probably require comparison at equal vehicle performance levels. Power-to-weight ratios and, hence, acceleration times and top speeds for the diesel vehicles cited above are inferior to values expected in typical gasoline-powered vehicles.

Gas Turbine, Stirling Cycle, and Rankine Cycle Engines

Gas turbine, Stirling cycle, and Rankine engines all employ steady flow or continuous combustion processes operated with fuel-lean overall mixture ratios. In a strict sense, the gas turbine is an internal combustion engine since high temperature combustion products serve as the cycle working fluid. Rankine and Stirling engines are external combustion devices with heat exchanged between high temperature combustion gases and the enclosed cycle working fluid.

In contrast to the situation with conventional spark ignition piston engines, the major obstacles related to use of continuous combustion power plants are in the areas of manufacturing costs, durability, vehicle performance, and fuel economy. The problem of exhaust emissions, which involves primarily the combustion process, has been less significant than the foregoing items.

As a consequence of lean combustion, these continuous combustion power plants are characterized by low HC and CO emissions. Several investigators have reported data indicating that existing combustion systems are capable of approaching or meeting statutory 1975 and 1976 vehicle emissions standards for HC and CO (26,27).

For a given power output, NO_x emissions appear to be lower than those of conventional uncontrolled gasoline engines. However, it has been shown that existing combustors probably will not meet the statutory 1976 NO_x standard when installed in motor vehicles (26).

The formation of NO_x in continuous-flow combustors has been found to result from the presence of high temperature zones with local fuel-air ratios in the vicinity of stoichiometric conditions. Approaches suggested for minimizing NO_x formation have involved reduction of these localized peak temperatures through such techniques as radiation cooling, water injection, and primary zone air injection. Other approaches include lean mixture primary zone combustion such that local maximum temperatures fall below levels required for significant NO formation. Laboratory gas turbine combustors employing several of these approaches have demonstrated the potential for meeting the 1976 standards (28). With a laboratory Stirling engine combustor, Philips has measured simulated Federal vehicle test procedure emissions levels well below 1976 statutory levels (29).

Conclusion

As an alternative to the conventional internal combustion engine equipped with exhaust treatment devices, modified combustion engines can, in principle, yield large reductions in vehicle exhaust emissions. Such modifications include stratified charge engines of both open and dual chamber design. On an experimental basis, prototype stratified charge engines have achieved low exhaust emissions with fuel economy superior to that of conventional engines controlled to similar emissions levels.

The diesel engine is capable of achieving low levels of light HC, CO, and NO_x emissions with excellent fuel economy. Potential problems associated with widespread diesel use in light-duty vehicles are initial cost, large engine size and weight for a given power output, the possibility of excessive particulate and odorant emissions, and excessive engine noise.

Several power plants based on continuous combustion processes have the potential for very low exhaust emissions. These include the gas turbine, the Rankine engine, and the Stirling engine. However, at the present time major problems in the areas of manufacturing costs, reliability, durability, vehicle performance, and fuel economy must be overcome. As a consequence, these systems must be viewed as relatively long range alternatives to the piston engine.

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Table I

Federal Exhaust Emissions Standards
Emissions, Grams/Mile[1]

	1975			1976			1977
		Interim			Interim		
	Statutory	U.S.	California	Statutory	U.S.	California	Statutory
HC	0.41	1.5	0.9	0.41	0.41	0.41	0.41
CO	3.4	15	9.0	3.4	3.4	3.4	3.4
NO_x	3.0	3.1	2.0	0.4	2.0	2.0	0.4

[1]As measured using 1975 CVS C-H procedure.

Table II

Average Emissions from Military Jeep Vehicles with Stratified Engine Conversions (Reference 10)

Engine	Miles	Emissions, g/Mile[2]			CVS Fuel Economy, mpg
		HC	CO	NO_x	
L-141 Ford[1] PROCO	Low 17,123	0.37 0.64	0.93 0.46	0.33 0.38	18.5-23
L-141 Texaco[1] TCCS	Low 10,000	0.37 0.77	0.23 1.90	0.31 0.38	16-22

[1]Engines equipped with oxidation catalysts and exhaust gas recirculation.

[2]1975 CVS C-H test procedure.

Table III

Average Low Mileage Emissions Levels - Ford PROCO Conversions (Reference 10)

	Emissions,[2] g/Mile			CVS Fuel Economy, mpg	Inertia Weight, Lb
	HC	CO	NO_x		
PROCO 141-CID[1] Capri Vehicles	0.12	0.46	0.32	20.4	2500
	0.13	0.18	0.33	25.1	
	0.11	0.27	0.32	22.3	
PROCO 351-CID[1] Torino Vehicle	0.30	0.37	0.37	14.4	4500
PROCO 351-CID[1] Montego Vehicles	0.36	0.13	0.63	-	-
	0.36	1.08	0.39	12.8	-

[1]All vehicles employed noble metal exhaust oxidation catalysts and exhaust gas recirculation.

[2]1975 CVS C-H test procedure.

Table IV

Honda Compound Vortex-Controlled Combustion-Powered Vehicle[1] Emissions (Reference 19)

	Emissions,[2] g/Mile			Fuel Economy, mpg	
	HC	CO	NO_x	1975 FTP	1972 FTP
Low Mileage Car[3] No. 3652	0.18	2.12	0.89	22.1	21.0
50,000-Mile Car[4] No. 2034	0.24	1.75	0.65	21.3	19.8

[1]Honda Civic vehicles.

[2]1975 CVS C-H procedure with 2000-lb inertia weight.

[3]Average of five tests.

[4]Average of four tests.

Table V

Emissions from Honda Compound Vortex-Controlled Combustion Conversion of 350-CID Chevrolet Impala (Reference 20)

Test	Emissions,[1] g/Mile			Fuel Economy, mpg
	HC	CO	NO_x	
1	0.27	2.88	1.72	10.5
2[2]	0.23	5.01	1.95	11.2
3[3]	0.80	2.64	1.51	10.8
4	0.32	2.79	1.68	10.2

[1]1975 CVS C-H procedure, 5000-lb inertia weight.

[2]Carburetor float valve malfunctioning.

[3]Engine stalled on hot start cycle.

Table VI

Steady State Emissions from Honda Compound Vortex-Controlled Combustion Conversion of 350-CID Chevrolet Impala (Reference 20)

Vehicle Speed, mph	Emissions, g/Mile					
	HC		CO		NO_x	
	350 CVCC	350 Stock	350 CVCC	350 Stock	350 CVCC	350 Stock
15	0.15	0.60	3.30	7.26	0.37	0.52
30	0.00	1.22	0.65	9.98	0.53	0.37
45	0.00	0.51	0.19	4.71	1.00	0.93
60	0.01	0.32	0.53	2.48	3.00	1.78

Table VII

Single-Cylinder Divided Combustion Chamber
Engine Emissions Tests (Reference 10)

Engine	NO_x Reduction Method	Emissions, g/ihp-hr			Fuel Economy, Lb/ihp-hr
		NO_x	HC	CO	
PROCO	EGR	1.0	3.0	13.0	0.377
Divided Chamber	None	1.0	0.4	2.5	0.378
PROCO	EGR	0.5	4.0	14.0	0.383
Divided Chamber	None	0.5	0.75	3.3	0.377

Table VIII

Automotive Diesel Engine Emissions
(Reference 25)

Vehicle	Emissions, g/Mile				Inertia Weight, Lb	Fuel Economy, mpg	
	HC (Cold Bag)	HC (Hot FID)	CO	NO_x		1975 FTP	1972 FTP
Mercedes 220DD	0.17	0.34	1.42	1.43	3500	23.6	23.3
Mercedes 220D (Modified)	0.13	0.23	1.08	1.48	3500	24.6	23.6
Opel Rekord 2100D	0.16	0.40	1.16	1.34	3000	23.8	23.2
Peugeot 504D	1.30	3.53	3.34	1.04	3000	25.2	24.2

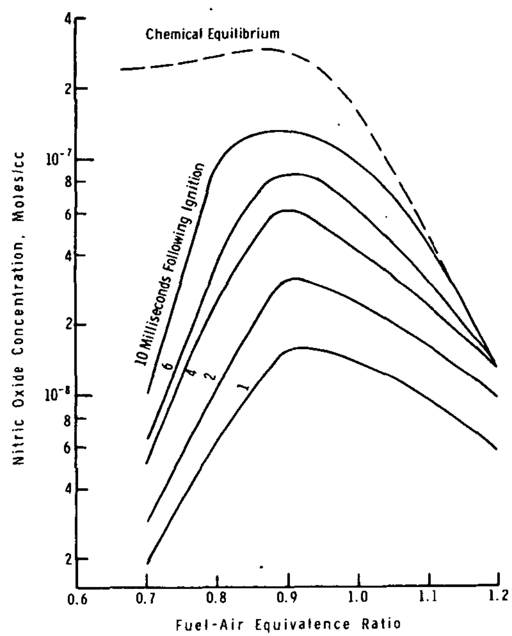

Figure 1: Rate of Nitric Oxide Formation in Engine Combustion Gases (Reference 6)

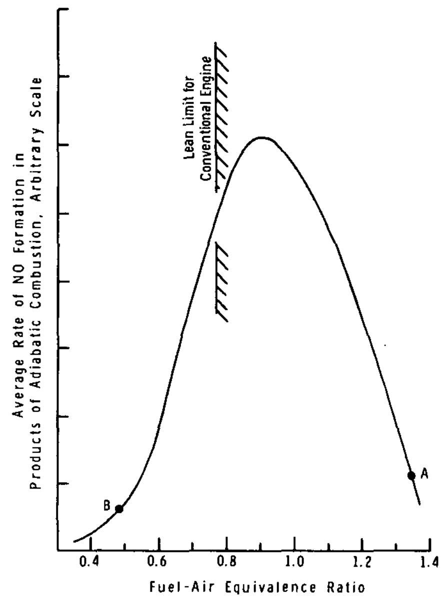

Figure 2: Influence of Fuel-Air Ratio on Rate of Nitric Oxide Formation

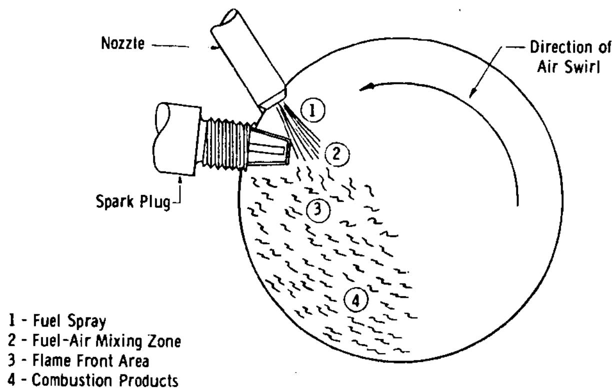

Figure 3: Texaco-Controlled Combustion System (TCCS)

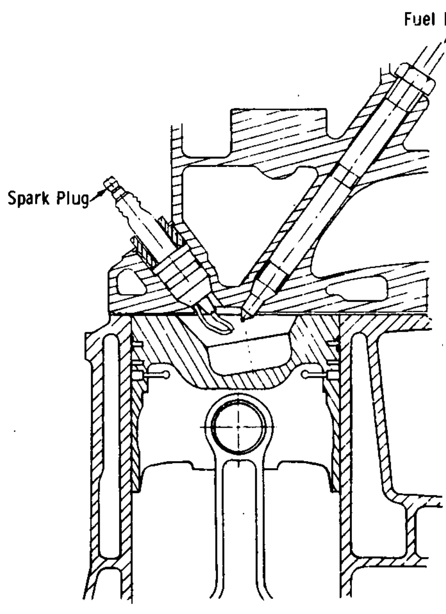

Figure 4: Ford-Programmed Combustion (PROCO) System

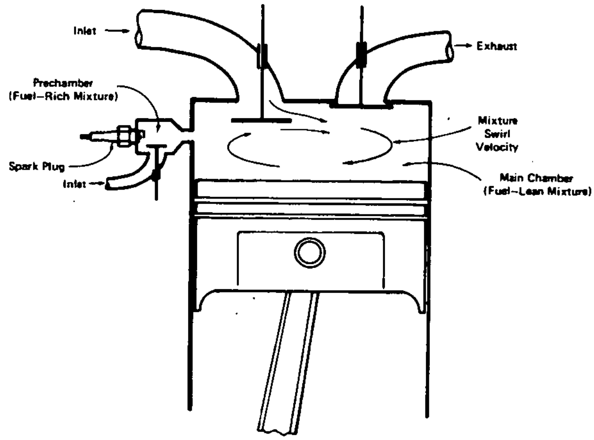

Figure 5: Schematic Representation of Prechamber Stratified Charge Engine

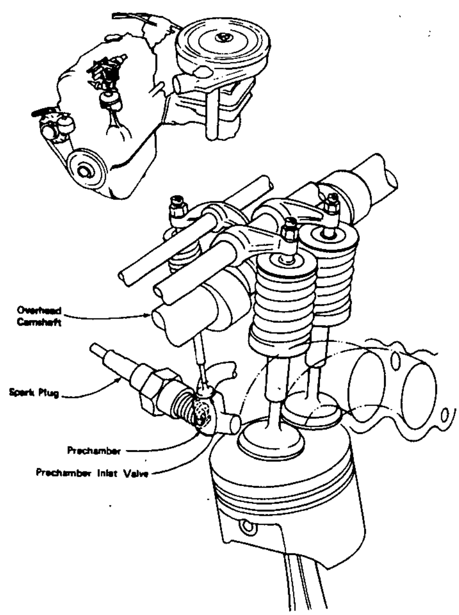

Figure 6: Honda CVCC Engine (Reference 19)

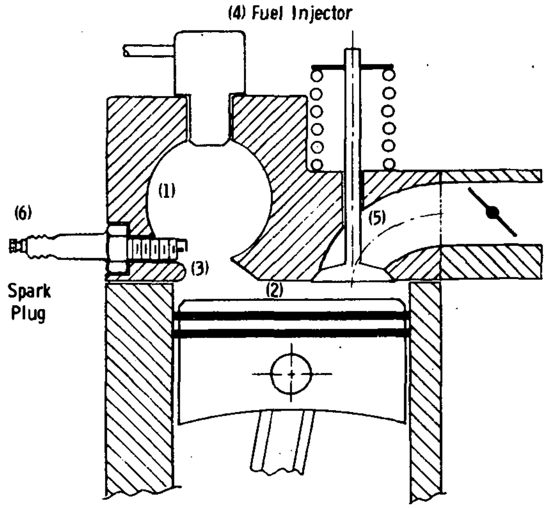

Figure 7: Schematic Representation of Divided Chamber Engine (Reference 21)

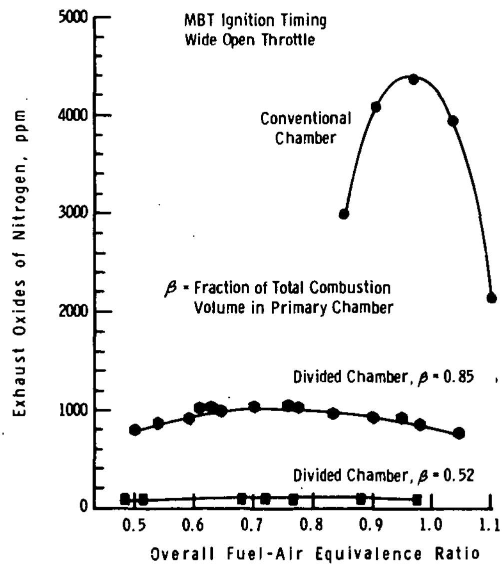

Figure 8: Comparison of Conventional and Divided Combustion Chamber NO_x Emissions (Reference 23)